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Serial Number of Application 09/782,897

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Number of Results returned (Minimum 50/ Maximum 300) 50

Keywords to emphasize

ART-UNIT: 376

09|782,897

PRIMARY-EXAMINER: Casaregola; Louis J.

ATTY-AGENT-FIRM: Miller; Albert J.

ABSTRACT:

A liquid fuel pressurization and control system is disclosed which utilizes either a helical flow pump, or a helical flow pump followed by a gear pump, to pressurize liquid fuel to precisely the pressure level required by a turbogenerator's combustor injectors. This eliminates the need to overpressurize the fuel then regulate the fuel pressure down using a flow control valve or a pressure control valve. The shaft torque and shaft speed of the pump are controlled by the turbogenerator's power controller so as to assure that the turbogenerator's speed is precisely controlled (e.g. within ten (10) rpm out of one hundred thousand (100,000) rpm), and that its turbine exhaust temperature is precisely controlled (e.g. within two (2) degrees Fahrenheit out of twelve hundred (1200) degrees Fahrenheit) over the full range of turbogenerator electrical output power. The system also provides cool, high pressure air to assist atomization of the liquid fuel in the injectors utilizing a variable speed helical flow compressor. The system also adjusts the relative fuel flow through the multiple fuel injectors to aid flame stability at low turbogenerator speeds and low output power levels.

19 Claims, 16 Drawing figures
Exemplary Claim Number: 1
Number of Drawing Sheets: 12

BRIEF SUMMARY:

TECHNICAL FIELD

This invention relates to the general field of turbogenerators and more particularly to an improved liquid fuel pressurization and control system for a turbogenerator.

BACKGROUND OF THE INVENTION

In a turbogenerator generating electricity and operating on a liquid fuel, it is necessary to increase the pressure of and atomize or vaporize the liquid fuel to be provided to the turbogenerator combustor. In addition, it is also desirable to increase the pressure of some of the turbogenerator compressor discharge air which is nominally supplied to the turbogenerator combustor and use this additionally compressed air to assist liquid fuel atomization in special fuel/air injectors used in the combustor. In order to have complete combustion, without the generation of undesirable combustion products such as COx and NOx, it is critical that the liquid fuel be completely atomized or vaporized when it enters the turbogenerator combustor. Further, if not fully atomized, the liquid fuel can leave varnish on any metal surfaces that it comes into contact with. The increased pressure liquid fuel and the increased pressure turbogenerator compressor discharge air (air assist air) can work together to accomplish complete atomization.

In addition, if the liquid fuel is at too high a temperature, the fuel injectors which deliver the liquid fuel to the turbogenerator can become vapor locked which will disrupt the continued flow of the liquid fuel to the combustor. It is, therefore, essential that the temperature of the liquid fuel be maintained below the temperature at which vapor lock can occur. Means to cool the liquid fuel may be required.

In a conventional turbogenerator operating on a liquid fuel, the speed of the turbogenerator is normally controlled by the interaction of liquid fuel flow rate and the load of the turbogenerator electrical output. Besides requiring a separate liquid fuel control and/or metering valve to regulate the liquid fuel flow rate, such a system requires a turbogenerator speed sensor, requires a turbogenerator turbine exhaust temperature sensor, is dependent upon turbogenerator load, would not be self-damping, and has certain inherent instabilities.

Further, in the operation of a turbogenerator, it has been difficult to sustain low power output operation. Inherently, the turbogenerator is designed for a continuous, steady-state, full power operation. When a low power output is required to be sustained, the fuel system does not inherently have the capability to adequately deal with this type of operation without some special measures being taken.

A new type of fuel pump and a new type of compressor to supply air assistance for fuel/air injectors appears to be warranted. Centrifugal pumps and compressors are potential candidates for both liquid fuel pressurization and control and for air compression used for fuel/air atomizing injectors. However, centrifugal pumps and compressors operate best (with high efficiencies) when they have a high throughput flow rate and a low pressure rise relative to their tip speed. These operating conditions are characterized as high specific-speed conditions. Under these conditions, a centrifugal compressor can operate with an efficiency on the order of seventy-eight percent (78%). But the flow rate and pressure rise requirements for fuel pressurization and air assist compression for the liquid fuel pressurization and control system are for low specific-speed compressors (low throughput flow rate and high pressure rise relative to the compressor's tip speed). A centrifugal pump and compressor operating under these conditions would have an efficiency of less than twenty percent (20%). Under these conditions it would require a very large number of centrifugal compressors in series (e.g. ten (10)) to produce the same pressure rise for a given tip speed as could one (1) helical flow compressor.

A helical flow compressor is a high-speed rotating machine that accomplishes compression by imparting a velocity head to each fluid particle as it passes through the machine's impeller blades and then converting that velocity head into a pressure head in a stator channel that functions as a vaneless diffuser. While in this respect a helical flow compressor has some characteristics in common with a centrifugal compressor, the primary flow in a helical flow compressor is peripheral and asymmetrical, while in a centrifugal compressor, the primary flow is radial and symmetrical. The fluid particles passing through a helical flow compressor travel around the periphery of the helical flow compressor impeller within a generally horseshoe shaped stator channel. Within this channel, the fluid particles travel along helical streamlines, the centerline of the helix coinciding with the center of the curved stator channel. This flow pattern causes each fluid particle to pass through the impeller blades or buckets many times while the fluid particles are traveling through the helical flow compressor, each time acquiring kinetic energy. After each pass through the impeller blades, the fluid particles reenter the adjacent stator channel where they convert their kinetic energy into potential energy and a resulting peripheral pressure gradient in the stator channel. The multiple passes through the impeller blades (regenerative flow pattern) allows a helical flow compressor to produce discharge heads of up to fifteen (15) times those produced by a centrifugal compressor operating at equal tip speeds. A helical flow compressor operating at low specific-speed and at its best flow can have efficiencies of about fifty-five percent (55%) with curved blades and can have efficiencies of about thirty-eight percent (38%) with straight radial blades.

A helical flow pump has the same basic design as a helical flow compressor.

Among the advantages of a helical flow pump or compressor or a helical flow turbine are:

(a) simple, reliable design with only one rotating assembly;

(b) stable, surge-free operation over a wide range of operating conditions (i.e. from fill flow to no flow);

(c) long life (e.g., 40,000 hours) limited mainly by their bearings;

(d) freedom from wear product and oil contamination since there are no rubbing or lubricated surfaces utilized;

(e) fewer stages required when compared to a centrifugal compressor; and

(f) higher operating efficiencies when compared to a very low specific-speed (high

head pressure, low impeller speed, low flow) centrifugal compressor.

The flow in a helical flow pump or compressor can be visualized as two fluid streams which first merge and then divide as they pass through the pump or compressor. One fluid stream travels within the impeller buckets and endlessly circles the pump or compressor. The second fluid stream enters the pump or compressor radially through the inlet port and then moves into the horseshoe shaped stator channel which is adjacent to the impeller buckets. Here the fluids in the two streams merge and mix. The stator channel and impeller bucket streams continue to exchange fluid while the stator channel fluid stream is drawn around the pump or compressor by the impeller motion. When the stator channel fluid stream has traveled around most of the compressor periphery, its further circular travel is blocked by the stripper plate. The stator channel fluid stream then turns radially outward and exits from the compressor through the discharge port. The remaining impeller bucket fluid stream passes through the stripper plate within the buckets and merges with the fluid just entering the compressor/turbine.

The fluid in the impeller buckets of a helical flow pump or compressor travels around the compressor at a peripheral velocity which is essentially equal to the impeller blade velocity. It thus experiences a strong centrifugal force which tends to drive it radially outward, out of the buckets. The fluid in the adjacent stator channel travels at an average peripheral velocity of between five (5) and eighty (80) percent of the impeller blade velocity, depending upon the pump or compressor discharge flow. It thus experiences a centrifugal force which is much less than that experienced by the fluid in the impeller buckets. Since these two centrifugal forces oppose each other and are unequal, the fluid occupying the impeller buckets and the stator channel is driven into a circulating or regenerative flow, The fluid in the impeller buckets is driven radially outward and "upward" into the stator channel. The fluid in the stator channel is displaced and forced radially inward and "downward" into the impeller bucket.

While the fluid in either a helical flow pump or compressor is traveling regeneratively, it is also traveling peripherally around the stator-impeller channel. Thus, each fluid particle passing through a helical flow pump or compressor travels along a helical streamline, the centerline of the helix coinciding with the center of the generally horseshoe shaped stator-impeller channel.

SUMMARY OF THE INVENTION

In the present invention, the liquid fuel pressurization and control system and method utilizes a pump whose speed and shaft torque directly controls the pressure of the liquid fuel delivered to the turbogenerator combustor. This method includes establishing the turbogenerator speed and turbogenerator turbine discharge gas temperature required based upon the power load requirements of the turbogenerator, establishing the liquid fuel pressure requirements to produce the established turbogenerator speed and temperature, and commanding the pump to produce the established liquid fuel pressure by controlling the speed or the torque of the pump.

The liquid fuel pressurization and control system for a turbogenerator includes a pump for supplying pressurized liquid fuel to the liquid fuel injectors of the turbogenerator combustor while the turbogenerator compressor supplies pressurized combustion air to the turbogenerator combustor. A motor, such as a permanent magnet motor, drives the compressor. A compressor motor inverter drive provides electrical power to the motor and receives operational speed and phase data from the motor. The inverter drive also receives torque and maximum speed control signals from the turbogenerator power controller which receives a speed feedback signal from the compressor motor inverter drive. A turbogenerator speed signal and a turbine exhaust gas temperature signal are provided to the turbogenerator power controller from the turbogenerator. A separate compressor can also be utilized to increase the pressure of the turbogenerator compressor discharge air to provide an air assist to the turbogenerator combustor nozzles and also to cool the liquid fuel. A helical flow compressor and pump can be utilized as the compressor and pump for the liquid fuel pressurization and for the air assist compression.

A helical flow compressor system is typically thirty (30) to forty (40) times smaller than systems with reciprocating compressors; consumes about one-third

(1/3) of the energy that other liquid fuel pressurization systems use; does not require the use of an accumulator; does not compress the liquid fuel to a pressure that is higher than is needed and then throw the extra pressure away through valve based flow or pressure regulation; does not cycle on and off; does not operate in a pulsed mode; and is very fast and responsive being controlled by the same computer that controls the entire turbogenerator combustion process.

The helical flow compressor is driven at high speed on the order of twenty four thousand (24,000) rpm by a permanent magnet motor/generator. It is designed to produce very high pressure for a given tip speed of impeller.

A conventional centrifugal pump takes liquid fuel such as diesel oil or gasoline and passes it through an impeller blade which imparts kinetic energy to the liquid fuel. That kinetic energy or velocity energy is converted to pressure energy in a diffuser channel. This happens once each time the liquid fuel goes through the pump. In order to obtain a large pressure rise, you either have to have a high speed impeller with a large diameter, or you have to have a large number of compression stages.

A helical flow pump or compressor also takes inlet liquid fuel or air into the impeller blade where it picks up kinetic energy or velocity energy and then the liquid fuel or air goes into a stator channel (which is in effect a vaneless diffuser) where the kinetic energy is turned back into pressure energy. While this happens only once in the typical centrifugal pump or compressor, it typically happens twelve (12) to fifteen (15) times in a helical flow pump or compressor. Thus, you can obtain about twelve (12) to fifteen (15) times as much pressure rise in a single stage of a helical flow pump or compressor as you can obtain in a single stage of a centrifugal pump or compressor.

The helical flow compressor is also designed to produce very low flows whereas the centrifugal compressor requires higher flows for greater efficiency. Because of this, centrifugal compressors operating at high flows have higher efficiencies than helical flow compressors running at their best efficiencies. When, however, you compare centrifugal compressors with helical flow compressors with the same low flows, helical flow compressors actually have higher efficiencies. A centrifugal compressor operating at its best operating condition would be operating at about a seventy eight percent (78%) efficiency. The centrifugal compressor would, however, be operating at its best flow which will be well above the flows needed by the turbogenerator. The helical flow compressor operating at its best flow can have efficiencies with curved blades of about fifty five percent (55%) and with straight blades of about thirty eight percent (38%). The efficiency of the helical flow compressor with straight blades for the flows required by the turbogenerator is about twenty five percent (25%) and with curved blades may be slightly over thirty percent (30%). On the other hand, the centrifugal compressor would be under twenty percent (20%) because it would be operating at such a low flow, well below where it is designed to operate at. At these low flows, there is a lot of scroll leakage losses in the centrifugal compressor.

The helical flow compressor has a lightweight wheel or impeller for a given throughput flow rate and pressure rise. The centrifugal compressor will be somewhat heavier with less ability to accelerate and decelerate than the helical flow compressor. If both a centrifugal compressor and a helical flow compressor are both designed to provide what the turbogenerator requires, the impeller of the helical flow compressor would be much lighter and much easier to accelerate and decelerate than the impeller of the centrifugal compressor.

Since the pressure of the liquid fuel introduced into the turbogenerator combustor is a function of the speed of the helical flow pump, the system computer can control the data which controls the motor which controls the pump and effectively has the computer control either the pressure or the flow of the helical flow pump which is pressurizing the liquid fuel. In a helical flow pump driven by a permanent magnet motor, or by an induction motor, you can control the torque the motor makes or control the speed or a mix of the two. Typically in this application, the torque is controlled since that controls the pressure rise of the compressor. The buckets have a known cross sectional area at a known radius to the center of the motor shaft. Thus, there is a given pressure rise for a given motor torque. The liquid fuel to the turbogenerator can therefore be effectively controlled.

After ignition, combustion generated turbine torque accelerates the turbogenerator which raises the pressure of the turbogenerator compressor. As the turbogenerator compressor increases the pressure of the combustion air, you will also need to increase the liquid fuel pressure to keep it somewhat higher so that there is a positive flow of liquid fuel to the combustor injectors. If for any reason the turbogenerator gets to a speed so as to produce more turbogenerator compressor discharge pressure than the liquid fuel pressure, the liquid fuel flow will stop and no liquid fuel will enter the turbogenerator combustor and the turbogenerator goes down in speed. This in fact constitutes a speed control mechanism which works extremely well.

A conventional liquid fuel pressurization and control system controls the fuel flow rate delivered to the turbogenerator but not the pressure of the fuel delivered to the turbogenerator. If the flow is held constant the turbogenerator speed can run away when the electric power load suddenly drops off. If the electrical load coming out of the turbogenerator drops off, more torque is available from the turbine to accelerate the wheel. The problem is controlling the speed in the system based upon the control of flow of liquid fuel. Only a high speed, high gain servosystem can prevent speed surges if fuel flow is controlled rather that fuel pressure.

In the present invention, the pressure rather than the mass flow of the liquid fuel is controlled and set to a pressure such as of twenty five (25) psi gauge. The turbogenerator will automatically accelerate if the compressor discharge pressure is twenty three (23) psi gauge. At that point, the turbogenerator is getting the amount of fuel it needs to run. With a drop off of load at the turbogenerator, the most that the turbogenerator speed can increase is that change in speed associated with an increase of two (2) psi in compressor discharge pressure. The speed goes up about three percent (3%) or four percent (4%) (considered to be a speed error) and stabilizes out as the gaseous fuel flow naturally drops down. Essentially what the computer based control logic does is reduce this small error by using a limited amount of gain or by using limited authority integration reducing this small error to essentially zero with small variations in fuel pressure. This makes a stable servocontrol.

With prior art technology, there is almost no gain in the turbogenerator by virtue of the fuel fluidics and the compressed air pneumatics, the gain is all in the computer that is controlling the liquid fuel and that's a hard thing to do. What is done in the present invention is to use the turbogenerator as a moderate gain servosystem on its own right. If you control the fuel pressure, you control the turbogenerator speed within a five percent (5%) tolerance range for a wide range of output power. The turbogenerator keeps itself from overspeeding and enables the system to get by with a very low gain (thus stable) servosystem that is computer based. Noting the power that the customer wants electrically, the computer goes to look-up tables to determine the speed and temperature at which the turbogenerator should be operating to produce that power. Another look-up table determines what pressure the liquid fuel should have to be consistent with that selected turbogenerator speed and temperature. The fuel pressure is then commanded to be equal to that level by changing the speed of the helical flow pump or by changing the torque of the helical flow pump motor. These conditions are obtained with a very small error because the prediction algorithms can be extremely accurate. A very small authority or limited gain integral proportional controller algorithm can trim out the last errors in speed, exhaust gas temperature, or output power.

A liquid fuel pressurization and control system based on the present invention stabilizes much faster than systems that over pressurize the fuel, then reduce the feel pressure with a mass flow control valve. It has been demonstrated that this system can control a turbogenerator over a speed range of twenty four thousand (24,000) rpm to ninety six thousand (96,000) rpm and can control the turbogenerator speed to within ten (10) rpm and that it can also control the turbine exhaust temperature to within two (2) degrees Fahrenheit. It is a very friendly system which does not overshoot and is capable of overcoming many of the difficulties of prior systems.

It is therefore a principle objective of the present invention to provide an improved liquid fuel pressurization and control system and method for a turbogenerator.

It is another objective of the present invention to provide a liquid fuel

pressurization and control system having means to pressurize liquid fuel to the precise pressure required by a turbogenerator combustor's injection nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that does not pressurize the liquid fuel to a pressure substantially above that required by a turbogenerator combustor's injection nozzles then subsequently subregulate that pressure down to the level required by the nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that does not require a mass flow control valve or a pressure control valve to assure that the liquid fuel is pressurized to the precise pressure required by a turbogenerator combustor's injection nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that utilizes a variable speed pump to both pressurize the liquid fuel and to control its pressure and flow to precisely match the requirements of the turbogenerator combustor's injection nozzles with no subsequent valve based subregulation.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that utilizes a variable speed pump having electrical power requirements much lower than those of an automotive fuel pump (owing to pumping only the pressure needed by the turbogenerator rather than overpumping then subregulating) and having this pump submersion mounted in the fuel tank.

It is another object of the present invention to provide a liquid fuel pressurization and control system having a compressor that does not have output flow rates or output pressures that pulsate.

It is another object of the present invention to provide a liquid fuel pressurization and control system having a compressor that does not have to be cycled on and off to control the average liquid fuel discharge pressure.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that utilizes either a helical flow pump, or a helical flow pump followed by a gear pump, to pressurize the liquid fuel to precisely the pressure required by a turbogenerator combustor's injection nozzles with no subsequent valve based subregulation.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that utilizes a pump that is integrated with a drive motor and mounts the rotating pump elements on the same shaft on which is mounted the motor rotor.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that utilizes a pump motor and motor inverter drive that can be commanded to produce a given motor/pump shaft torque or can be commanded to produce a given shaft speed and in any case will provide a signal that is a function of shaft speed. Note that if the pump motor is a d.c. motor (shunt or otherwise) this objective can be met with interdependent control of motor current and voltage.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that, through fuel control, can control turbogenerator speed to within 10 rpm over an operating speed range of zero to 100,000 rpm and can control turbine exhaust temperature to within 2 degrees Fahrenheit over an operating range of 300 degrees Fahrenheit to 1200 degrees Fahrenheit for the entire output power range.

It is another objective of the present invention to provide a liquid fuel pressurization and control system having means to supply cool air at up to 6 psig above turbogenerator centrifugal compressor discharge pressure to assist atomization of the liquid fuel in the turbogenerator combustor's air assisted injector nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system utilizing a variable speed compressor to further

pressurize some of the air from the turbogenerator centrifugal compressor discharge in order to supply cool air at up to 6 psig above turbogenerator compressor discharge pressure to assist atomization of the liquid fuel in the turbogenerator combustor's air assisted injector nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system utilizing a variable speed helical flow compressor to further pressurize some of the air from the turbogenerator centrifugal compressor discharge in order to supply cool air at up to 6 psig above turbogenerator compressor discharge pressure to assist atomization of the liquid fuel in the turbogenerator combustor's air assisted injector nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that varies the speed of the helical flow air atomization assist compressor to provide adequate but not excessive air for atomization and adequate but not excessive air for fuel cooling in the injector nozzles (to prevent vapor lock) without utilizing excess electrical motor/inverter power.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that can reduce the fuel flow through some of the liquid fuel injector nozzles under conditions of low turbogenerator speed and low turbogenerator combustion temperature in order to stabilize the combustion flame, avoid flameouts and reduce the turbogenerator idle speed and idle fuel consumption.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that can reduce the fuel flow through some of the liquid fuel injector nozzles under conditions of low turbogenerator speed and low turbogenerator combustion temperature utilizing solenoid shut-off valves that are sequentially activated for each injector nozzle based on turbogenerator speed and/or turbogenerator turbine exhaust temperature and/or fuel flow rate.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that can reduce the fuel flow through some of the liquid fuel injector nozzles under conditions of low turbogenerator speed and low turbogenerator combustion temperature utilizing a proportional valve or multiple proportional valves that have their flow conductances adjusted as a function of turbogenerator speed and/or turbogenerator turbine exhaust temperature and/or fuel flow rate.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that can reduce the fuel flow through some of the liquid fuel injector nozzles under conditions of low turbogenerator speed and low turbogenerator combustion temperature utilizing a flexure valve or multiple flexure valves that have their flow conductances adjusted as a function of fuel pressure. These flexure valves use no solenoid, use no electrical power, require no conditioning and control circuitry. They are controlled and powered solely by the pressure of the liquid fuel used by the injector nozzles.

It is another objective of the present invention to provide a liquid fuel pressurization and control system that controls the torque and speed of the utilizes either a helical flow pump, or a helical flow pump followed by a gear pump, to pressurize the liquid fuel to precisely the pressure required by a turbogenerator combustor's injection nozzles with no subsequent valve based subregulation.

DRAWING DESCRIPTION:

BRIEF DESCRIPTION OF THE DRAWINGS

Having thus described the present invention in general terms, reference will now be made to the accompanying drawings in which:

FIG. 1 is a plan view of a turbogenerator set utilizing the liquid fuel pressurization and control system and method of the present invention;

FIG. 2 is a perspective view, partially cut away, of a turbogenerator for the

LITIGATION SEARCH FOR LINDA SHOLL: US 5,873,235 (reissue 09/782,897)

Files searched in QUESTEL ORBIT:
Cluster : LEGAL
Databases : LGST (legal status), CRXX (claims legal status), PAST (patent status), LITA (LitAlert)

?us5873235/pn

LGST	1
CRXX	1
PAST	2
LITA	0

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1/4 LGST (1/1) - (C) LEGSTAT
PN  - US 5873235  [US5873235]
AP  - US 990467/97  19971215  [1997US-0990467]
DT  - US-P
ACT - 19971215  US/AE-A
      APPLICATION DATA (PATENT)
      {US 990467/97  19971215  [1997US-0990467]}
    - 19990223  US/A
      PATENT
    - 19990706  US/CC
      CERTIFICATE OF CORRECTION
UP  - 1999-27

2/4 CRXX (1/1) - (C) CLAIMS/RRX
AN  - 3113424
PN  - 5,873,235 A  19990223  [US5873235]
PA  - Capstone Turbine Corp
PT  - M (Mechanical)
ACT - 20010213  REISSUE REQUESTED
      ISSUE DATE OF O.G.: 20010717
      REISSUE REQUEST NUMBER: 09/782897
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      Reissue Patent Number:

UP  - 1999-26
UACT- 2001-07-17

3/4 PAST (1/2) - (C) PAST
AN  - 200129-001168
PN  - 5873235  A  [US5873235]
DT  - A (UTILITY)
OG  - 2001-07-17
CO  - REA
ACT - REISSUE APPLICATION FILED
SH  - REISSUE APPLICATION FILED

4/4 PAST (2/2) - (C) PAST
AN  - 199927-000220
PN  - 5873235  A  [US5873235]
DT  - A (UTILITY)
OG  - 1999-07-06
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CO - COR
ACT - CERTIFICATE OF CORRECTION
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Files searched in LEXIS and NEXIS:
All patent files

PATNO IS 5873235

Your search request has found 1 PATENT through Level 1.

LEVEL 1 - 1 OF 1 PATENT

<5,873,235>

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Feb. 23, 1999

Liquid fuel pressurization and control method

REISSUE: Reissue Application filed Feb. 13, 2001 (O.G. Jul. 17, 2001) Ex. Gp.: 3746; Re. S.N. 09/782,897

CORE TERMS: turbogenerator, compressor, liquid fuel, helical, pump, pressurization, fuel, stator, fluid, impeller...
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File searched: CASES

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File searched in DIALOG:
File 345:Inpadoc/Fam.& Legal Stat 1968-2001/UD=200130
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S1 1 PN="US 5873235"

1/39/1
DIALOG(R)File 345:Inpadoc/Fam.& Legal Stat

14364646
Basic Patent (No,Kind,Date): US 5752380 A 19980519 <No. of Patents: 002>
Patent Family:

Patent No	Kind	Date	Applic No	Kind	Date

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    US 5752380      A    19980519    US 730941       A    19961016  (BASIC)
    US 5873235      A    19990223    US 990467       A    19971215
Priority Data (No,Kind,Date):
    US 730941 A  19961016
    US 990467 A  19971215
    US 730941 A3 19961016

PATENT  FAMILY:
UNITED STATES OF AMERICA (US)
  Patent (No,Kind,Date):  US 5752380  A   19980519
    LIQUID FUEL PRESSURIZATION AND CONTROL SYSTEM (English)
    Patent Assignee:  CAPSTONE TURBINE CORP  (US)
    Author (Inventor):  BOSLEY ROBERT W  (US); EDELMAN EDWARD C  (US);
      LAMPE STEVEN W  (US); MILLER RONALD F  (US)
    Priority (No,Kind,Date):  US 730941  A   19961016
    Applic (No,Kind,Date):  US 730941  A   19961016
    National Class: *  060039281; 060734000
    IPC: *  F02C-009/26
    Language of Document:  English
  Patent (No,Kind,Date):  US 5873235  A   19990223
    LIQUID FUEL PRESSURIZATION AND CONTROL METHOD (English)
    Patent Assignee:  CAPSTONE TURBINE CORP  (US)
    Author (Inventor):  BOSLEY ROBERT W  (US); EDELMAN EDWARD C  (US);
      LAMPE STEVEN W  (US); MILLER RONALD F  (US)
    Priority (No,Kind,Date):  US 990467  A   19971215; US 730941  A3
      19961016
    Applic (No,Kind,Date):  US 990467  A   19971215
    Addnl Info:  5752380  Patented
    National Class: *  060039030
    IPC: *  F02C-009/26
    Derwent WPI Acc No: *  G  99-179599; G  99-179599
    Language of Document:  English

UNITED STATES OF AMERICA (US)
  Legal Status (No,Type,Date,Code,Text):
    US 5752380      P    19961016  US AE       APPLICATION DATA (PATENT)
                              (APPL. DATA (PATENT))
                              US 730941  A   19961016
    US 5752380      P    19961016  US AS02     ASSIGNMENT OF ASSIGNOR'S
                              INTEREST
                              CAPSTONE TURBINE CORPORATION 6025 YOLANDA
                              AVE. TARZANA, CALIFORNIA 91356 ; BOSLEY,
                              ROBERT W. : 19961016; EDELMAN, EDWARD C. :
                              19961016; LAMPE, STEVEN W. : 19961016;
                              MILLER, RONALD F. : 19961016
    US 5752380      P    19980519  US A        PATENT
    US 5873235      P    19961016  US AA       PRIORITY
                              US 730941  A3  19961016
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                              (APPL. DATA (PATENT))
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    US 5873235      P    19990223  US A        PATENT
    US 5873235      P    19990706  US CC       CERTIFICATE OF CORRECTION

                    END LITIGATION SEARCH US 5873235

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